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# Boundary layer control as a method of gas turbine blade cooling

Ness, Dwight Osten

St. Paul, Minnesota; University of Minnesota

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BOUNDARY LAYER CONTROL AS A METHOD  
OF GAS TURBINE BLADE COOLING

A THESIS

Submitted to the Graduate Faculty  
of the  
University of Minnesota

by  
DWIGHT C. <sup>Ten</sup>NESS  
COMDR. U.S.N.

In Partial Fulfillment of the Requirements  
for the  
Degree of Master of Science  
in  
Aeronautical Engineering

August  
1949

TABLE 1. INCREASED SOCIAL TRANSFER  
DURING 1960-1961, U.S.

TABLE 1.

TABLE 1. INCREASED SOCIAL TRANSFER

TABLE 1.

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TABLE 1.

TABLE 1. INCREASED SOCIAL TRANSFER

TABLE 1.

TABLE 1.

## ACKNOWLEDGMENTS

The author wishes to express his sincere appreciation to the following who aided in this study:

Professors B. J. Robertson, W. A. Hall, T. M. Murphy, and K. E. Neumeier of the Mechanical Engineering Department, for their assistance, advice and suggestions.

Shop personnel of the Oak Street Laboratory for their assistance in construction of the test equipment.



ACKNOWLEDGMENTS

The author wishes to express his sincere appreciation to the following who aided in this study:

Professor E. J. Robertson, R. A. Hall, L. E. Kopp, and E. H. Kessler of the Statistical Engineering Department, for their assistance, advice and suggestions.

Miss Margaret of the Oak Street Laboratory for their assistance in maintenance of the test equipment.

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## OBJECT AND SCOPE

The object of this thesis was to determine the feasibility of cooling gas turbine blades by introduction of a controlled boundary layer of cool air over the blade surface.

This investigation included a static test of a single instrumented turbine blade in a variable high velocity, high temperature gas stream with variable cooling air flow. Two configurations of the test blade were used to produce variation in boundary layer control.

# APPENDIX

The object of this study was to determine the feasibility of making gas turbine blades by injection of a controlled boundary layer of air over the blade surface.

This investigation included a study of a single laminated injection blade in a turbine stage velocity, high temperature gas turbine with certain modifications. The modification of the test blade was used to produce results in boundary layer control.

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## INTRODUCTION

Maximum effort in the development of gas turbines is being exerted to improve specific power output, to reduce specific fuel consumption and to increase reliability. The most promising field for the attainment of these objectives lies in increasing the turbine inlet temperature which is presently limited by permissible operating temperatures of blading materials. An investigation of the gas turbine thermodynamic cycle reveals the magnitude of improvement possible by increasing turbine operating temperatures. Such an investigation conducted by the NACA (Ref. 1) shows that for a given mass flow of working fluid the specific power output is proportional and the specific fuel consumption is inversely proportional to the inlet temperature. Fig. 1 illustrates this relation.

The increase of turbine inlet temperature, however, is limited by high temperature strength of blade materials. The development of high temperature metals is proceeding, but at a slow rate. How slowly metallurgical progress has been made is shown in Fig. 83 of Ref. 2. Allowable blade temperatures advanced from 1180° F. in 1935 to 1200° F. by 1940 and to 1385° F. by 1945. The rate of increase has been no greater since 1945.

Non-metallic materials such as ceramics have yet to demonstrate their adaptability to the rigorous service





requirements of turbine blading. As a result the use of some method of cooling the gas turbine blading presents itself as the method of allowing higher gas temperatures with present materials.

Several methods of blade cooling have been proposed and evaluated. A discussion of these methods as related to this thesis follows.

Late model German turbojet engines such as the Juno 004 employed hollow turbine blade cooled internally by means of air blown into the root and exhausted at the tip. 1650° F. turbine inlet temperature was used with 7% of compressor air output required to cool blades approximately 400° F.

The NACA has proposed (Ref. 1) an improvement to this method by inserting a core in the blade, leaving a small annular air passage. It was found that the heat transfer from blade to cooling air was principally in the boundary layer and adjacent cooling air so the insert permitted similar cooling with less air flow. Fig. 2 graphs the results of this improvement.

Another cooling method consists of circulating water through internal passages in the blade. This system of cooling has accomplished very large blade temperature reductions. German applications (Ref. 3) conducted by Dr. Schmidt permitted 850-930° F. blade temperatures with a gas temperature of 2200° F. Fig. 3 shows an NACA analytical



regulation of engine speed, as a result the use of  
 some method of cooling the gas turbine during its  
 life as the method of allowing slight gas temperature rise  
 without restriction.

Several methods of blade cooling have been pro-  
 posed and evaluated. A discussion of these methods is pre-  
 sented in this paper.

The first method proposed consists of the  
 use of air which enters the turbine blades directly by  
 means of the blade tip. The air is drawn in at the  
 tip of the blade and is forced into the blade passage  
 by the centrifugal force. This method is not very effective  
 because the air is not cooled before it enters the blade  
 passage.

The second method proposed (Ref. 1) is to inject air  
 into the blade passage by means of a hole in the blade, forcing  
 the air into the passage. It was found that the air  
 transfer from the blade to the passage air was relatively low  
 because of the high gas temperature. The air was cooled by  
 means of a heat exchanger before it entered the blade  
 passage. This method is more effective than the first  
 method.

The third method proposed (Ref. 2) consists of forcing  
 air through internal passages in the blade. This system  
 of cooling has been investigated very extensively and  
 is the most effective method of cooling. Various systems (Ref. 3) consisting of  
 internal passages, external passages, and a combination of  
 both have been investigated. The most effective system is  
 the one which combines internal and external passages.

investigation of water cooling which also gave considerable blade temperature reduction. It must be pointed out that while water cooling is very effective, the problems of handling the high temperature, high pressure water flow at high rates makes service application of this method difficult.

A modification of the foregoing system, known as rim cooling, has also been tried. Here water is circulated through the rotor rim so as to extract heat from the blade root. Less temperature reduction is obtained and the disadvantages of the water coolant system still exist. Fig. 4 shows rim cooling effectiveness. It can be seen from the figure that having a blade material of high thermal conductivity,  $K_M$ , is essential to this method.

All of the foregoing methods employ the same basic principle of cooling. They do not inhibit the heat transmission to the blade, but do increase the internal conductivity, or removal of heat, thereby affecting cooling. The proposal of this thesis is to substitute a boundary layer of cool air over the blade's surface in order to inhibit the heat transmission from the hot gases to the blade.

A coating of high temperature ceramic of low conductivity would embody this same principle. Fig. 5 shows the effectiveness of ceramic coatings of various thicknesses. While this is a very promising field insofar as temperature of operation is concerned, the inherent defects of brittle-

Investigation of water cooling which also gave considerable  
 of the comparative reduction. It must be pointed out that  
 while water cooling is very effective, the results of  
 feeding the high temperature, high pressure water from a  
 high water source involve attention at this point also.  
 Well.

A modification of the foregoing system, known as  
 the cooling, has also been tried. Here water is circulated  
 through the tower and is so heated that from the glass  
 front. Such temperature reduction is obtained and the dis-  
 advantages of the water cooling system will be noted. It  
 shows the cooling effectiveness. It can be seen from the  
 figure that using a single system of high thermal capacity  
 water, it is essential to this method.

All of the foregoing methods employ the same  
 basic principle of cooling. They do not include the heat  
 transmission in the liquid, but do require the internal  
 conductivity, or transfer of heat, thereby affecting cooling.  
 The purpose of this device is to establish a boundary  
 layer of cool air over the liquid's surface in order to re-  
 duce the heat transmission from the hot gases to the liquid.  
 A cooling of high temperature liquids of low vis-  
 cosity could easily be accomplished. With a more  
 the effectiveness of various systems of various substances.  
 While this is a very practical field in the laboratory  
 of operation is concerned, the present status of this



ness, thermal shock sensitivity and low tensile strength have obviated service use of ceramic covered blading.

If a region of low thermal conductivity can be interposed between the gas and the blade, then the objective of blade cooling could be accomplished. The natural boundary layer on the blade is such a region. However, the natural boundary layer forms at the temperatures of the gas. In this experiment the use of relatively cool air from the engine compressor is suggested to form a lower temperature boundary layer.

The justification of this idea is based on one of the fundamental laws of heat transfer, Faurier's equation for conduction (Ref. 4). (Experience has shown radiation effects to be secondary). Stated mathematically for steady state conduction:

$$q = KA \frac{dt}{dx}$$

where  $q$  = rate of heat transfer.

$K$  = coefficient of thermal conductivity.

$A$  = crosssectional area of path.

$\frac{dt}{dx}$  = temperature gradient in direction of heat flow per unit distance.

This law shows that for a given configuration the rate of heat transfer from gas to blade may be made by reducing  $K$  and/or  $\frac{dt}{dx}$ .

$K$  for air is reduced by reducing temperature.

This is shown mathematically from Lucheng equation:

heat, thermal conductivity and low thermal capacity

have obtained results on the various properties listed.

If a region of low thermal conductivity can be

interposed between the gas and the blade, then the objective

of blade cooling could be accomplished. The natural tendency

of any layer on the blade is to form a porous structure, and

natural porous layer forms at the temperature of the gas.

In this experiment the use of relatively small air flow the

region considered is suggested by the fact that porous

porous layer.

The construction of this test is based on the use of

the fundamental laws of heat transfer, Fourier's equation

for conduction (Eq. 4), (convection) and from radiation

effects to be determined. (Detailed experimentally for steady

state conditions)

$$q = \frac{k}{L} \Delta T$$

where  $q$  = rate of heat transfer.

$k$  = coefficient of thermal conductivity.

$L$  = thickness of wall of plate.

$\Delta T$  = temperature gradient in direction of heat

flow per unit distance.

This law shows that for a given configuration the

rate of heat transfer from gas to blade may be made by re-

$$q = h A (T_g - T_b)$$

where  $h$  is the heat transfer coefficient by convection.

This is shown mathematically from Fourier's equation



$$K = K_{32} \frac{492 + C}{T + C} \left( \frac{T}{492} \right)^{3/2}$$

where  $T$  = absolute temperature

$C$  = constant (.0129 for air).

$K_{32}$  =  $K$  at 32° F.

The temperature gradient from the boundary layer to blade,  $\frac{dt}{dx}$ , is reduced by the use of the cool air controlled boundary layer. In fact, the cool air boundary layer will at first be lower in temperature than the blade so that the blade will transfer heat to the boundary layer. However, the temperature gradient from the hot gas to the boundary layer would be increased so it would be rapidly heated. The optimum configuration might therefore require a series of bleeds from the blade so the average temperature of the layer along the blade would be minimized.

In the author's experience a controlled boundary layer has been successfully employed to cool a liquid rocket nozzle. In 1936 the author collaborated in the construction of a liquid rocket motor in which a boundary layer of coolant air bled into the nozzle enabled prolonged operation. The nozzle was of mild steel yet endured the very high temperature rocket exhaust gases better than any contemporary nozzles of superior materials.

1918 and 1919, 2 specimens \* 2

[illegible]

in the machine's operation a similar condition  
never has been successfully repeated to such a degree  
before. In 1955 the engine exhibited in the com-  
parison of a light engine test in which a steady  
layer of coolant air did not the same stable pressure  
operation. The reason for this was not known at  
very high temperatures under which the engine  
operated under a similar condition.

## TEST EQUIPMENT

Fig. 5 shows the complete test equipment layout schematically. The test blade was mounted in a closed test section supplied with hot gas from a single J-33 combustion chamber. Fig. 6 is a photograph of the test section mounted on the burner. Air was supplied to the burner from the compressor of a naturally aspirated Allison V-1710 engine, Fig. 7. The quantity of gas flow was regulated by throttling the Allison engine while its temperature was controlled by burner fuel pressure. The temperature of the blade was measured by two thermocouples. All control and measurement was done from the control panel adjacent to the gas turbine test cell. Fig. 8 is a photograph of the control panel. The air which formed the controlled boundary layer was supplied from the laboratory air main at regulated pressure. The quantity of cooling air was measured in a standard design sharp edged orifice meter, shown in Fig. 9.

The test blade was manufactured from a solid Juno 004 turbine blade. Availability was the reason for selection of this blade. The "tinidur" type alloy (30% nickel, 14% chrome, 1.75% titanium, 12% carbon, balance, iron) possessed very difficult machining properties and low thermal conductivity. The blade roots were cut off flat for convenient mounting and the tip shortened by 3/4 inch



# TEST EQUIPMENT

Fig. 3 shows the complete test equipment layout.

essentially. The test stand was mounted in a closed steel  
enclosure supplied with air from a single 1-1/2 inch  
diameter. Fig. 4 is a photograph of the test stand enclosure.

on the burner. Air was supplied to the burner from the  
compressor of a specially designed system 7-1/2 inch  
diameter. The quantity of air flow was regulated by means

with the Allen engine valve for temperature was controlled  
by means of a pressure. The temperature of the flame was  
measured by two thermocouples. All control and measurement

was done from the control panel adjacent to the gas turbine  
test cell. Fig. 5 is a photograph of the control panel.  
The air which formed the controlled boundary layer was supplied

from the laboratory air main at regulated pressure.  
The quantity of heating air was measured in a standard 1-1/2  
inch square orifice water meter, shown in Fig. 6.

The test stand was constructed from a solid  
steel base plate. Availability was the reason for  
selection of this stand. The "standard" type alloy steel

material, 1/2 inch, 1.75 inch, 1.5 inch, 1.25 inch, 1.0 inch, 0.75 inch,  
from standard very difficult working properties and the  
standard construction. The blade cross was cut off the

for convenient mounting and the tip diameter by 1/2 inch



because of space limitations in the test section.

For the first test, configuration A was manufactured. In this blade the cooling air supply hole was drilled up the blade from root to  $1/4$  inch of tip through the thickest section. This hole was .20 inch in diameter. The air bleed holes ( $1/16$  inch) were drilled from blade surfaces joining the supply hole. They were placed at a 45 degree angle with blade surface. There were six bleed holes to each surface. The exits were ground out with a fish tail countersink pattern to distribute the bleed air spanwise. A .15 inch hole was drilled up the leading edge for location of the thermocouple tip at midspan. The trailing edge was too thin to permit similar treatment so a  $3/32$  inch hole was drilled chordwise at midspan to snugly hold a thermocouple bead. The thermocouple was lead in in a stainless steel tube one inch downstream and bent 90 degrees and cemented in the trailing edge hole. Figs. 12, 13 and 14 show the thermocouple mounting. Fig. 10 shows the A blade between a standard blade and a shortened standard blade. The boundary layer is introduced at approximately the 30% chord point.

Configuration B blade is shown in Fig. 11. The boundary layer air supply was introduced through a .15 inch drilled passage  $1/4$  inch behind the leading edge. A 60 degree included angle slot was milled down the length of the leading edge and  $3/32$  inch bleed holes drilled joining the

Source: U.S. Census Bureau, *Marriage, Divorce, Remarriage in the 1990s*, p. 10.

For the first test, configuration 1 was used:

100-443887-100

Called by the name of the person who called.

the oldest region. The only one, 20 km in diameter,

The air blood gases of the blood were similar

without joining the supply unit. This was done at a

15 degrees north with 1000 ft. clouds.

1. The first step is to identify the problem or question that needs to be answered. This involves understanding the context and the specific requirements of the task.

See David and staff/agents at various interviews 11/11/03

approximately 3.16 with bolts are drilled up the loading axis

for located at the University of the Pacific. The Wells-

ing data can be used to identify specific areas for improvement.

These data are consistent with the hypothesis that the observed effects are due to the presence of a large, stable, and well-organized social network.

a characteristic of the system. The characteristic of the system is a

—el DE 2000 han desarrollado una gran variedad de actividades

STRAUS AND JACOBSON IN THE TWENTY-THIRD CENTURY

It was 10 years ago that we started this company. We are now

the 3 trials between a standard finish and a shortened finish.

and finally, the secondary layer is subjected to a second

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Received 21. 2. 2009; accepted 10. 3. 2009

United States, 65 into United States, 65

These included supply and delivery from the January to the

loading edge and 20% from black color driver (tested on



supply passage. In this design the cooling enters the slot opposed by stagnation pressure and flows out of the slot on both edges, forming the boundary layer.

The test section consisted of the blade mounting block shown in Fig. 13, and two side plates made of six inch channel. The bottom was closed with a 1/2 inch plate so that the hot gases which entered at the top were constrained to exhaust through the open side. The entrance and exit dimension are 3.5 x 4.5 inches. Fig. 12 pictures the complete test section. Fig. 14 shows another view of the blade mounting block. The test blade is centrally located with two parallel mounted standard blades to guide the flow.

Temperatures of the test blade were measured by 20 gauge chromel-alumel thermocouples which read on a Brown recorder. The small size thermocouples provided fast response and use of standard sillimanite insulations in the blade. The insulators were ground slightly in diameter for mounting in blade. The thermocouple tips were firmly seated in 3/32 inch holes for maximum sensitivity to blade temperatures. The trailing edge thermocouple bead was buried completely to insure it would sense blade, rather than gas temperatures.

Compensated lead wires connected the thermocouples to the selector switch for the recorder to eliminate errors in readings by variation in ambient temperature. The gas

supply channels. In this design the venting system for the  
applied by atmospheric pressure and lines out of the area in  
both edges, located the secondary layer.

The test section consisted of the blade mounting

blades shown in Fig. 1, and the blade design was of the  
last design. The bottom was closed with a 1/2 inch plate  
so that the hot gases were not exposed to the air and con-  
sidered as exhaust through the open side. The exhaust  
and exit diameter was 1.5 x 4.0 inches. The 1/2 inch plate  
the complete test section. The 1/2 inch plate was at  
the blade mounting block. The test blade is completely iso-  
lated with the previous mounted standard blades to reduce the  
flow.

The purpose of the test blade was designed to  
the power characteristics characteristics which were as a three  
velocity. The test blade characteristics were tested in a  
space and was at standard atmospheric conditions in the  
blade. The test blade was placed slightly in diameter for  
mounting in blade. The aerodynamic data were then tested  
in 1/2 inch plate for various velocities in blade design.  
The trailing edge characteristics were not tested con-  
sidered to be same as the test blade, which was the

characteristics.  
The test blade was placed in the aerodynamic  
to the test section for the test section in standard three  
in velocity of velocity in standard aerodynamic. The test



temperature entering the test section,  $T_4$ , was measured by a radiation shielded chromel-alumel thermocouple.

temperature between the two points, 11.5 and 12.5, was 0.5°C. A radiation shielded thermocouple was used.

The thermocouple was calibrated against a standard.

The thermocouple was used to measure the temperature of the

liquid in the container and the temperature of the container

was measured by a thermocouple. The thermocouple was used to

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The thermocouple was used to measure the temperature of the

liquid in the container and the temperature of the container

## TEST PROCEDURE

The temperature of the test blade was read with and without cooling air flow under exactly similar flow conditions. This technique permitted comparison of the two temperatures obtained to show the blade reduction due to cooling air alone.

The Allison engine was first started and its speed set to obtain the desired flow rate of burner air. The flow rate was measured by means of the pressure drop across the orifice in the compressor inlet duct.

Next, combustion was initiated in the burner with the spark and acetylene flare and fuel pressure adjusted until desired gas temperature obtained. When conditions stabilized temperatures and pressures were recorded. During runs with cooling air, the flow rate was varied in increments of 1/10 inch of water and temperature recorded when stabilized.

In order to investigate the cooling effects over a broad range of gas velocities three settings of the Allison engine were used to give low, medium and high gas flow rates. However, the low flow rate is not included in this report as it was unrealistically low compared with actual turbine operation. The flow rate was below minimum idling rate for a J-33 engine.

# TEST RESULTS

The temperature of the test blade was read with

and slight cooling air flow under exactly similar flow conditions. This temperature provided comparison of the test temperatures obtained to show the blade position was in cooling air alone.

The silicon engine was fired started and the speed was adjusted to obtain the desired flow rate of burner air. The flow rate was measured by means of the pressure drop under the orifice in the combustion inlet duct.

Heat, combustion was obtained in the burner with the speed and cooling air flow and fuel pressure adjusted until desired gas temperature obtained. Then readings obtained temperatures and pressure were recorded. During flow with cooling air, the flow rate was varied in the amounts of 1/10 inch of water and temperature recorded when stabilized.

In order to investigate the cooling effects over a broad range of gas velocities three settings of the silicon engine were used to give low, medium and high gas flow rates. However, the low flow rate is not included in this report as it was unacceptably low compared with other service operation. The flow rate was raised almost three times for a high engine.



## TEST RESULTS

The results of the experiment are contained in tables I and II and the graphs, Figs. 16 through 19. The graphs are plotted to show temperature reduction versus weight of cooling air flow.

These graphs are similar in shape and show that the reduction in blade temperature was approximately twice as great in the leading edge as the trailing edge. This is to be expected because of the increase in boundary layer temperature resulting from heat transmission from the gas. Also, the thinness of the blade section near the trailing edge offers more resistance to heat flow internally.

The graphs also show that the temperature reduction rate is greatest (the slope is maximum) at low cooling air rates. This is evidence that the boundary layer is established at low flow rates and is effective in reducing heat transfer. Beyond a flow rate of .2 lb./min. most of the graphs become straight line functions. This apparently results from thickening of the boundary layer and shows the insulating effect is proportional to the thickness. This effect conforms with Fourier's law. The cooling effectiveness, particularly at low flow rates, is greater with this method than the method of Fig. 2.

# THE RESULTS

The results of the experiment are contained in  
Tables I and II and the graphs, Figs. 10 through 12. The  
graphs are plotted in three temperature reduction versus  
weight of cooling air flow.

These graphs are similar in shape and show that  
the reduction in flame temperature was approximately twice  
as great in the trailing edge as the leading edge. This  
is to be expected because of the increase in boundary layer  
thickness resulting from heat transmission from the gas.  
Also, the reduction of the flame velocity near the trailing  
edge offers more resistance to heat flow internally.

The graphs also show that the temperature reduction  
rate is greatest (the slope is greatest) at the leading  
edge air intake. This is evidence that the boundary layer  
is established at the flow intake and is effective in re-  
ducing heat transfer. Beyond a flow rate of 2 ft./min.  
most of the graphs show slightly less reduction. This  
apparently results from thinning of the boundary layer  
and shows the trailing effect is proportional to the  
thickness. This effect coincides with Fourier's law. The  
cooling effectiveness, particularly at low flow rates, is  
greater with this material than any material of Fig. 1.

To illustrate the cooling effectiveness consider the J-33 turbojet engine. Maximum cooling of 280 F. at 1600 F. gas temperature could be accomplished with only 2% of compressor air.

Configuration A produced more uniform results than those of configuration B, as can be seen by comparing Figs. 16 and 17 with 18 and 19. Configuration A curves plotted more parallel and gave results proportional to gas temperature, while configuration B curves intersect and are out of order with gas temperature increments. Configuration A probably gave more uniform boundary layer formation, since the flows were convergent rather than opposed. It was calculated that the stagnation point on the leading edge would fall in the milled slot so the cooling air would spill over both surfaces of the blade and form good boundary layers. From the non-uniform results at different flow rates stagnation point shifting may be indicated. Also, turbulence in the test section may have prevented uniform boundary layer formation and promoted mixing. It is believed that had the blade been manufactured with boundary layer control slots of the type used in airplane practice, much higher quality results would have been obtained. This type bleed was considered but discarded because of the machining problems, which would have required machining beyond shop capacity.



The illustration shows the cooling characteristics of the

the 1-25 inch diameter, 1000 ft. long, 1000 ft. diameter

1000 ft. long, 1000 ft. diameter, 1000 ft. long, 1000 ft. diameter

of the cooling air.

Consequently, the cooling air is not only

the cooling air is not only the cooling air, but it is also

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Uncontrollable variation in the speed of the Allison engine contributed to inaccuracy of data. A "hunting" of about fifty RPM occurred during much of the running, which produced 500 RPM variations in compressor speed. The variation in burner air flow caused drifting of gas temperatures.

Comparison of Figs. 16 with 17, and 18 with 19, shows cooling effectiveness variation with gas flow rate. Cooling effectiveness is greater at the lower flow rate. This is consistent with the laws of heat transmission by convection. The heat transfer from gas to blade increases with velocity.

From all graphs, the blade temperature reduction is shown to increase with gas temperature. This, also, is compatible with the laws of heat transfer, the  $\frac{dt}{dx}$  term of Fourier's equation increases so more heat is transferred to the outer boundary layer. But, the heat transfer to the blade through the laminar sublayer is of such small magnitude that the net effect is greater blade temperature reduction.

temperature variation in the speed of the

Alison's speed variation in the speed of the

"Alison" of about 15% and the speed of the

Alison, which produced 100% variation in the speed

speed. The variation in the speed of the Alison

is the variation.

Temperature of the Alison is 15% and is with 15.

Alison's speed variation in the speed of the

Alison's speed variation in the speed of the

Alison is consistent with the speed of the Alison

variation. The speed variation in the speed of the

Alison is consistent.

From all stages, the Alison's speed variation

is shown to be consistent with the speed of the

Alison's speed variation in the speed of the

Alison's speed variation in the speed of the

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## CONCLUSIONS

In view of the limited scope of the experimental tests, no detailed quantitative conclusions can be drawn. However, the results obtained from the foregoing tests do support the following general conclusions:

1. The introduction of a boundary layer of relatively cool air on a turbine blade in a high temperature, high velocity gas stream inhibits the transmission of heat from the gas to the blade, more than through the natural boundary layer, and results in reduction of blade temperature.

2. The magnitude of the reduction in blade temperature is proportional to the weight flow of air introduced into the boundary layer up to the limit investigated of 2% of the gas flow in an equivalent full scale engine.

3. This method of blade cooling is feasible insofar as weight flow of cooling air required to accomplish useful blade temperature reduction is concerned.



# EXPERIMENTAL

In view of the limited scope of the experimental tests, an detailed quantitative determination can be drawn. However, the results obtained from the foregoing tests do support the following general conclusions.

1. The introduction of a secondary layer of this highly pure air on a turbine blade in a high temperature, high velocity gas stream results in the formation of a thin film on the blade, with this layer the surface becoming lighter, and results in reduction of blade temperature.

2. The reduction of the reduction in blade temperature is proportional to the weight loss of air introduced into the boundary layer up to the limit specified of all of the film in the experiments will be indicated.

3. This method of blade cooling is feasible insofar as weight loss of cooling air relative to aerodynamic blade temperature reduction is concerned.

# OBSERVED TEST DATA

TABLE I

## HIGH AIR FLOW RUNS

COMPRESSOR RPM - 24,000

— GAS TEMPS —

COOLING AIR		800°F				1000°F				1200°F				1400°F				1600°F				CONFIGURATION
ΔP "H <sub>2</sub> O	W <sub>a</sub>	T <sub>BLE</sub>	T.R.	T <sub>STE</sub>	T.R.	T <sub>BLE</sub>	T.R.	T <sub>STE</sub>	T.R.	T <sub>BLE</sub>	T.R.	T <sub>STE</sub>	T.R.	T <sub>BLE</sub>	T.R.	T <sub>STE</sub>	T.R.	T <sub>BLE</sub>	T.R.	T <sub>STE</sub>	T.R.	
0	0	785		785		995		995		1190		1195		1390		1390		1585		1585		RUN 1
.1	.217	720	65	760	25	920	75	965	30	1110	80	1150	45	1255	135	1325	65	1440	145	1615	70	
.2	.345	710	75	755	30	920	75	965	30	1070	120	1135	60	1255	135	1325	65	1440	145	1505	80	
.3	.471	700	85	750	35	895	100	950	45	1050	140	1135	60	1220	170	1310	80	1385	200	1480	105	
.4	.600	690	95	735	50	875	120	945	50	1030	160	1110	85	1200	140	1300	90	1350	235	1465	120	
.5	.726	675	110	730	55	855	140	937	58	1015	175	1115	80	1185	205	1290	100	1335	250	1450	135	
.6	.846	665	120	730	55	850	145	930	65	1015	175	1115	80	1160	230	1270	100	1320	265	1450	135	
0	0	755		755		945		975		1150		1150		1325		1330		1480		1475		RUN 2
.1	.217	725	30	750	5	915	30	965	10	1080	70	1120	30	1300	25	1318	12	1350	130	1440	35	
.2	.345	695	60	735	20	900	45	955	20	1075	75	1120	30	1230	75	1310	20	1330	150	1440	35	
.3	.471	670	85	720	35	875	75	945	30	1042	108	1120	30	1200	125	1310	20	1310	170	1445	50	
.4	.600	650	105	720	35	855	90	935	40	1035	115	1120	30	1160	165	1295	35	1300	180	1425	50	
.5	.726	640	115	715	40	825	120	920	55	1000	150	1115	35	1135	190	1290	40	1270	210	1420	55	
.6	.846					800	145	905	70					1110	215	1275	55	1255	225	1420	55	
P <sub>f</sub>	PSI	800°F				1000°F				1200°F				1400°F				1600°F				
W <sub>f</sub>	lb/hr	74				93				107				129				146				
ΔP <sub>B.A.</sub>	"Hg	1.83				1.82				1.80				1.65				1.50				
W <sub>B.A.</sub>	lb/SEC	2.85				2.845				2.83				2.71				2.59				
P <sub>3</sub>	"Hg	14.8				15.0				16.1				17.1				16.8				
P <sub>E3</sub>	"Hg	15.4				15.4				16.5				17.5				17.2				
P <sub>4</sub>	"Hg	.72				.64				.60				.52				.50				
P <sub>E4</sub>	"Hg	11.45				11.8				13.1				13.75				13.8				
g	"Hg	10.63				11.16				12.5				13.23				13.3				
M <sub>4</sub>		.731				.75				.795				.82				.823				

TEMP: COOLING AIR - 80°F  
 AMBIENT (TEST CELL) 120°F

PRESS: ATMOS - 29.82 "Hg  
 AMBIENT (TEST CELL) 27.50 "Hg





# OBSERVED TEST DATA MEDIUM BURNER AIR FLOW RUNS COMPRESSOR RPM - 13,250

-16-

TABLE II

COOLING AIR ΔP "H <sub>2</sub> O W <sub>a</sub>		TEMP - 300°F				TEMP - 1000°F				TEMP - 1200°F				TEMP - 1400°F				TEMP - 1600°F					
		T <sub>BLE</sub>	T <sub>R</sub>	T <sub>STE</sub>	T <sub>R</sub>	T <sub>BLE</sub>	T <sub>R</sub>	T <sub>STE</sub>	T <sub>R</sub>	T <sub>BLE</sub>	T <sub>R</sub>	T <sub>STE</sub>	T <sub>R</sub>	T <sub>BLE</sub>	T <sub>R</sub>	T <sub>STE</sub>	T <sub>R</sub>	T <sub>BLE</sub>	T <sub>R</sub>	T <sub>STE</sub>	T <sub>R</sub>		
0	0	777		785		995		1000		1115		1200		1310		1345		1535		1510		RUN 3	CONFIGURATION A
.1	.217	720	57	750	35	912	95	103	31	1052	143	1135	65	1210	120	1338	51	1410	15	1550	40		
.2	.345	710	67	750	35	815	100	150	50	1042	150	1121	73	1230	100	1345	70	1422	103	1540	50		
.3	.471	645	82	750	35	860	135	140	60	1035	160	1120	80	1202	188	1312	83	1388	117	1550	40		
.4	.600	685	12	750	35	850	145	140	60	1015	180	1120	80	1175	215	1303	72	1365	220	1550	40		
.5	.726	610	107	750	35	835	160	140	60	1010	185	1120	80	1150	240	1300	75	1340	245	1550	40		
.6	.846	655	122	750	35	825	170	140	60	985	210	1120	80	1135	255	1300	75	1315	270	1540	50		
.7	.972	650	127	750	35	815	180	140	60	975	220	1120	80	1120	270	1300	75						
.8	1.09	645	132	750	35	800	175	130	70	965	230	1120	80										
0	0	720		725		730		745		1142		1147		1325		1320		1440		1480		RUN 4	CONFIGURATION B
.1	.217	645	75	706	11	860	170	730	15	1042	100	1120	21	1138	137	1245	35	1385	105	1455	25		
.2	.345	640	80	700	25	850	130	725	20	1030	112	1120	21	1182	143	1235	35	1365	105	1455	25		
.3	.471	632	88	700	25	831	133	725	20	1015	127	1110	31	1175	150	1275	45	1336	104	1450	30		
.4	.600	625	75	700	25	825	205	725	20	1000	142	1100	41	1120	205	1270	50	1275	215	1438	40		
.5	.726	605	115	675	30	800	230	715	30	965	111	1075	52	1103	217	1255	61	1250	240	1425	55		
.6	.846																						
.7	.972																						
.8	1.09																						

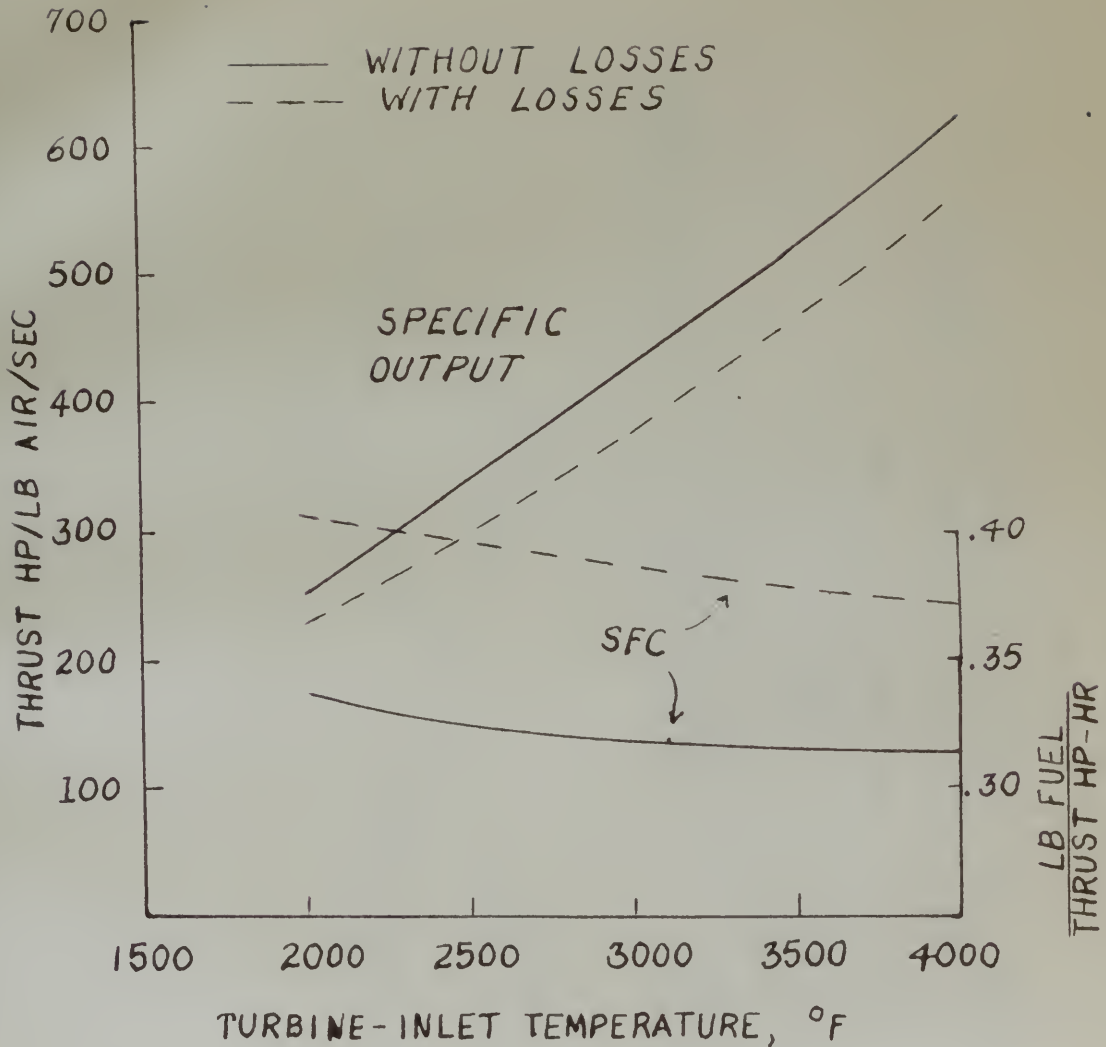
INSUFFICIENT COMPRESSED AIR SUPPLY

		800°F	1000°F	1200°F	1400°F	1600°F
P <sub>t</sub>	PSI	47	61	67	73	79
W <sub>t</sub>	lb/hr	47	65	80	90	103
ΔP <sub>B.A.</sub>	"H <sub>2</sub> O	1.05	.75	.88	.83	.80
W <sub>B.A.</sub>	lb/sec	2.16	2.06	1.48	1.42	1.89
P <sub>3</sub>	"H <sub>2</sub> O	7.1	7.8	8.4	8.6	9.1
P <sub>T3</sub>	"H <sub>2</sub> O	8.5	8.0	8.6	8.1	7.3
P <sub>4</sub>	"H <sub>2</sub> O	.8	.55	.4	.32	.30
P <sub>T4</sub>	"H <sub>2</sub> O	5.8	5.3	6.55	6.45	7.35
Δ	"H <sub>2</sub> O	5.3	5.25	6.15	6.00	7.05
M <sub>t</sub>		.417	.500	.551	.545	.60

TEMP: COOLING AIR 80°F PRESS: ATMOSPHERIC 21.82 "H<sub>2</sub>O

AMBIENT TEST CELL 120°F AMBIENT TEST CELL 27.50 "H<sub>2</sub>O





TURBOPROP ENGINE PERFORMANCE WITH +WITHOUT COOLING  
LOSSES. AIRPLANE SPEED, 500 MPH: MACH  
NO., 0.69; ALTITUDE 30,000 FT. (Ref. 1)

Fig. 1





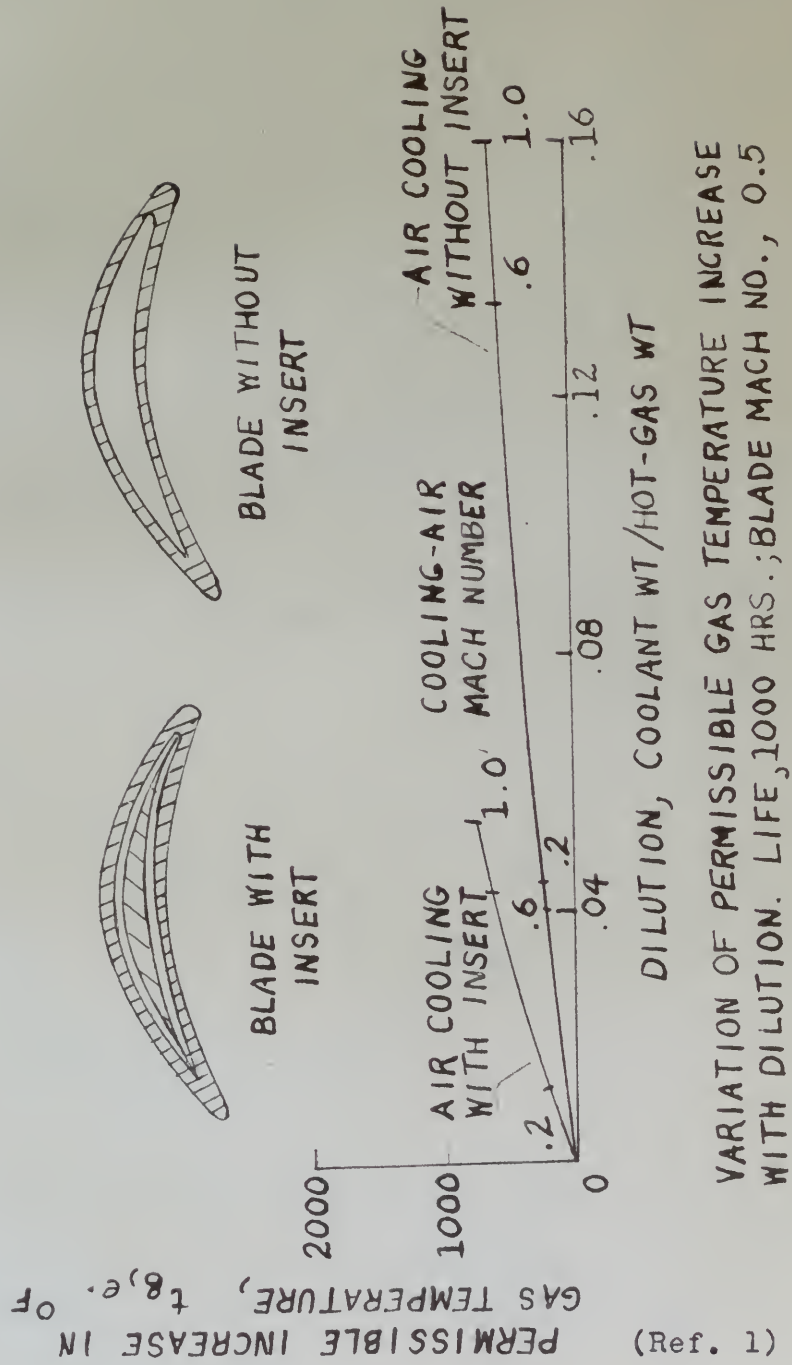
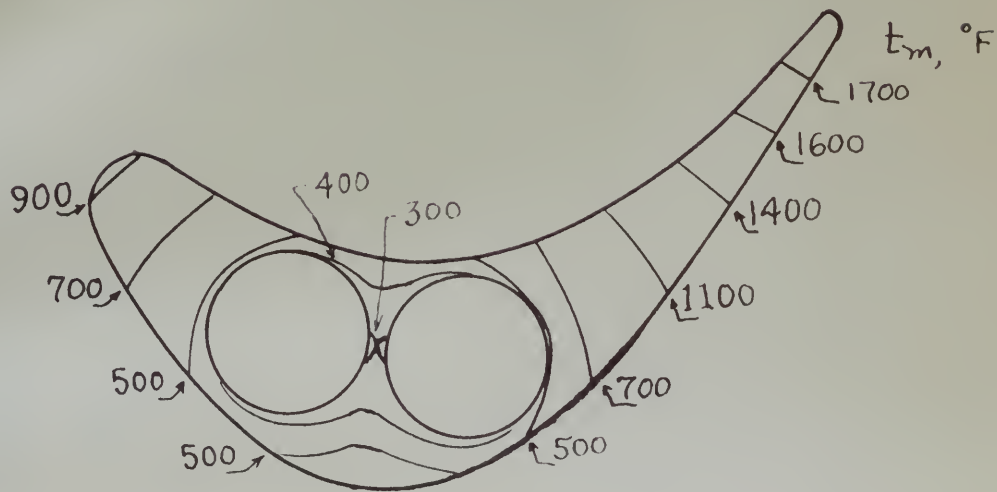


Fig. 2

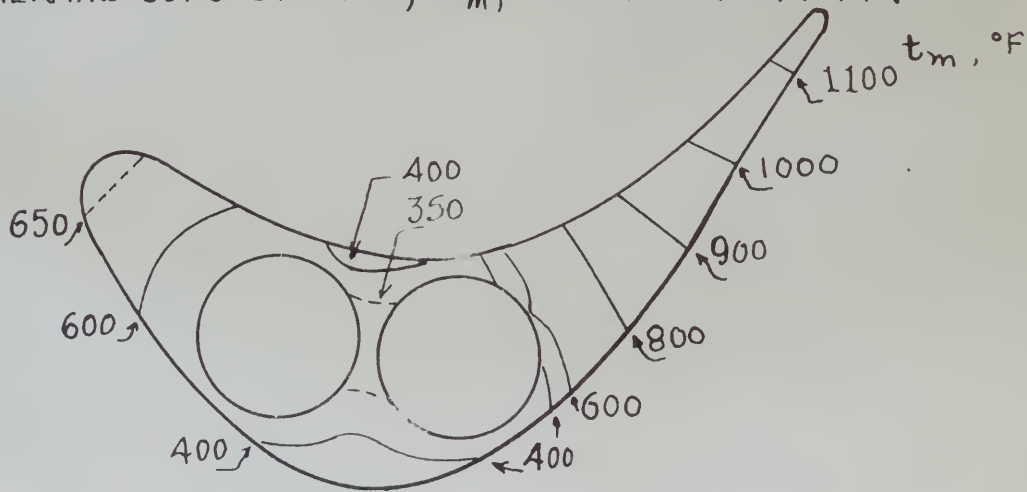




THERMAL CONDUCTIVITY,  $k_m$ , 15 BTU/(HR)(°F)(FT)



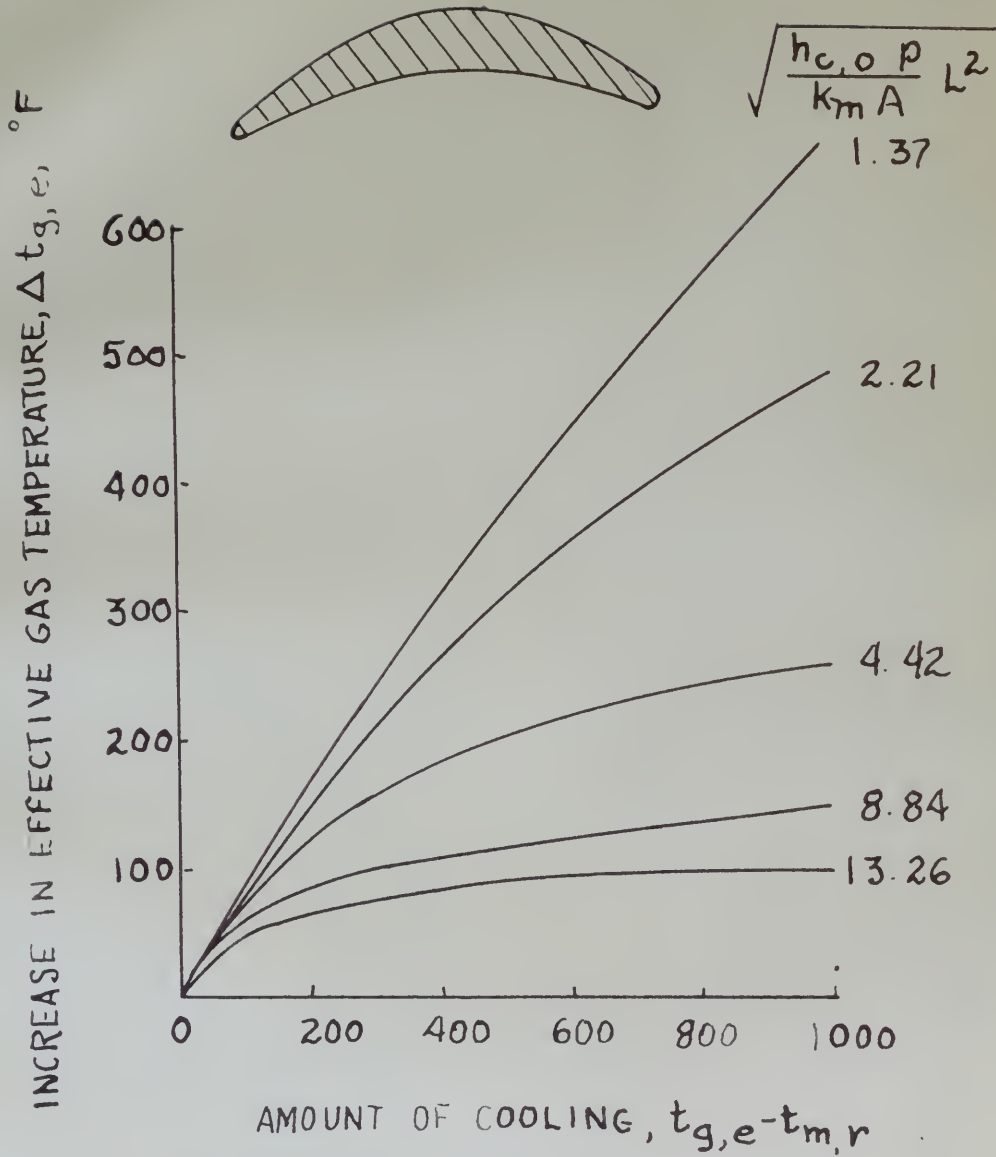
THERMAL CONDUCTIVITY,  $k_m$ , 100 BTU/(HR)(°F)(FT)



ISOTHERMS IN BLADE SECTIONS OF DIFFERENT CONDUCTIVITY MATERIAL WITH LIQUID COOLING. GAS FLOW, 55 LB/SEC; WATER FLOW, 6.42 LB/SEC; GAS TEMPERATURE, 2000° F; WATER TEMPERATURE, 200° F.  
(Ref. 1)

Fig. 3



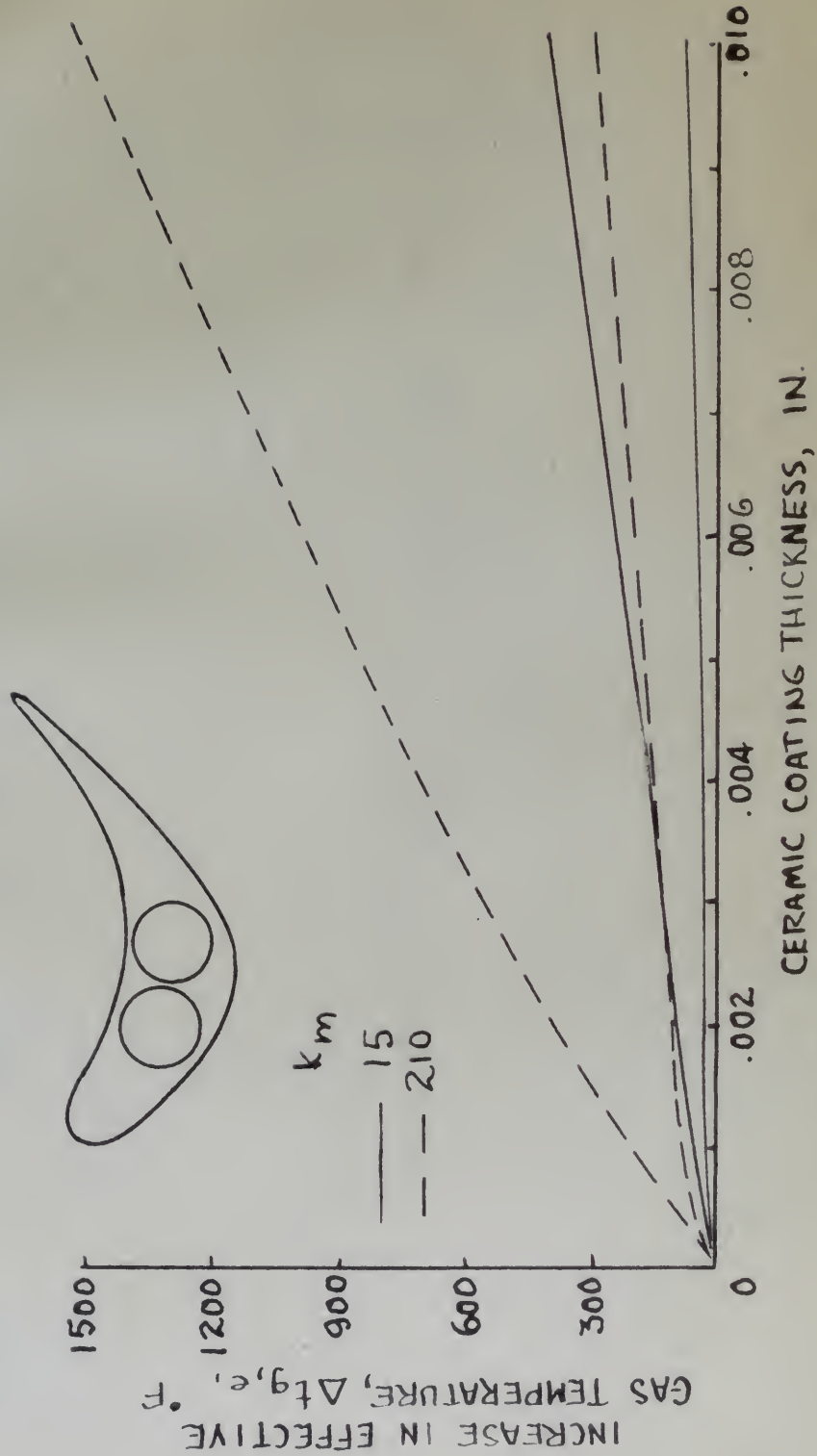


VARIATION OF RIM COOLING EFFECTIVENESS.  
 MAXIMUM ALLOWABLE MACH NUMBER, 0.5.  
 (Ref. 1)

Fig. 4







(Ref. 1)

Fig. 5

VARIATION OF INCREASE IN EFFECTIVE GAS TEMPERATURE WITH CERAMIC COATING THICKNESS FOR TWO METAL AND CERAMIC THERMAL CONDUCTIVITIES.





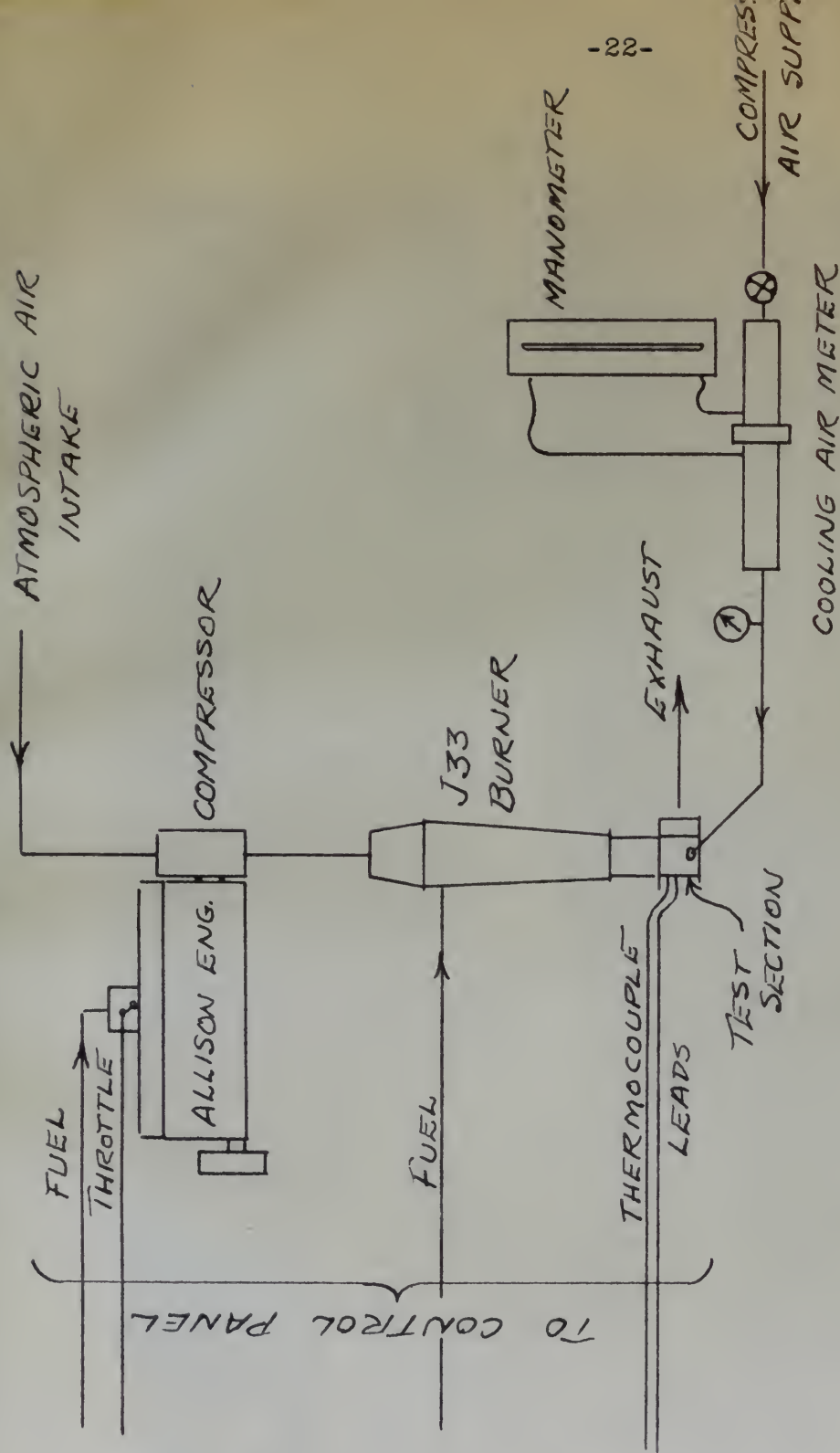
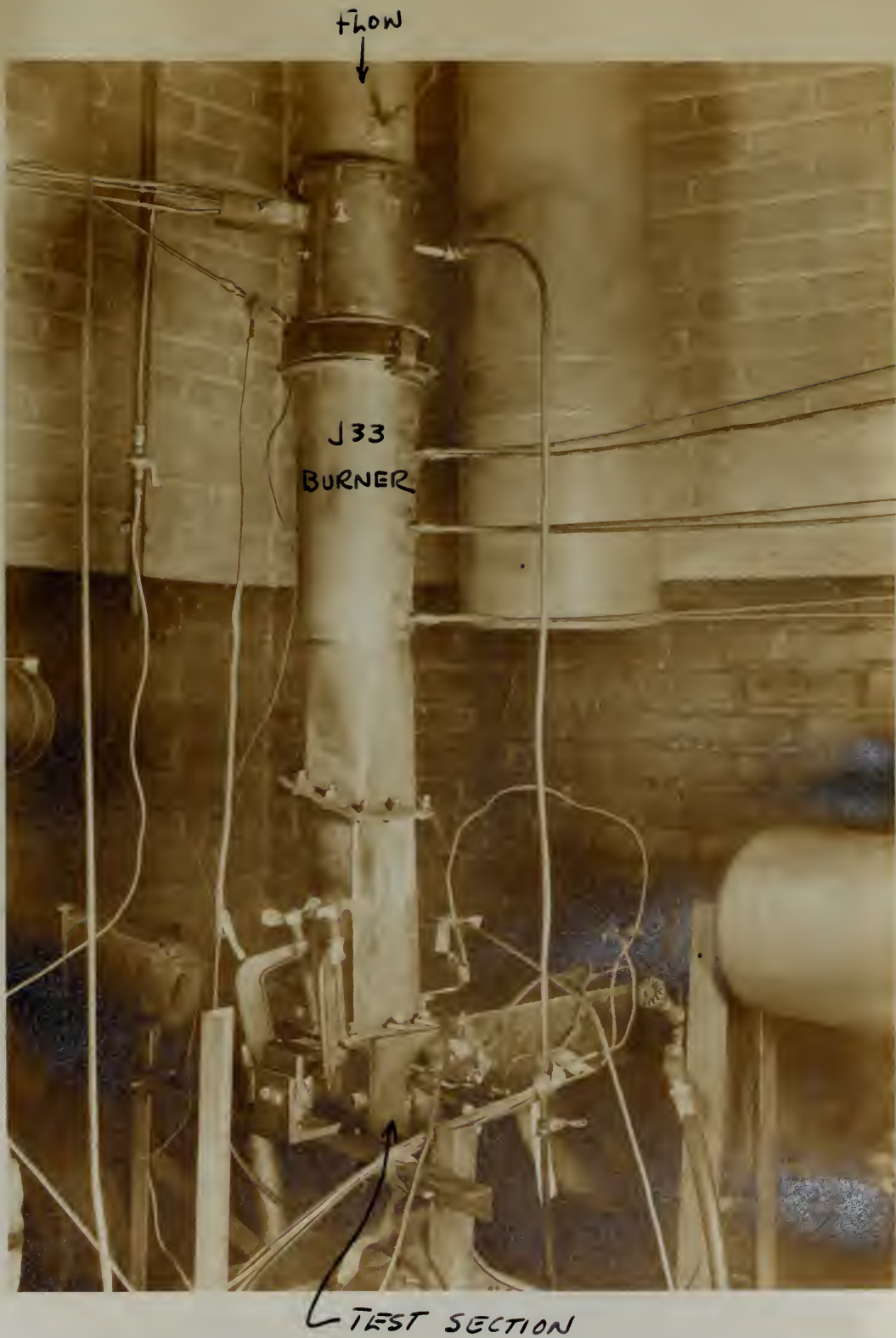


Fig. 6

SCHEMATIC DIAGRAM OF COMPLETE TEST LAYOUT

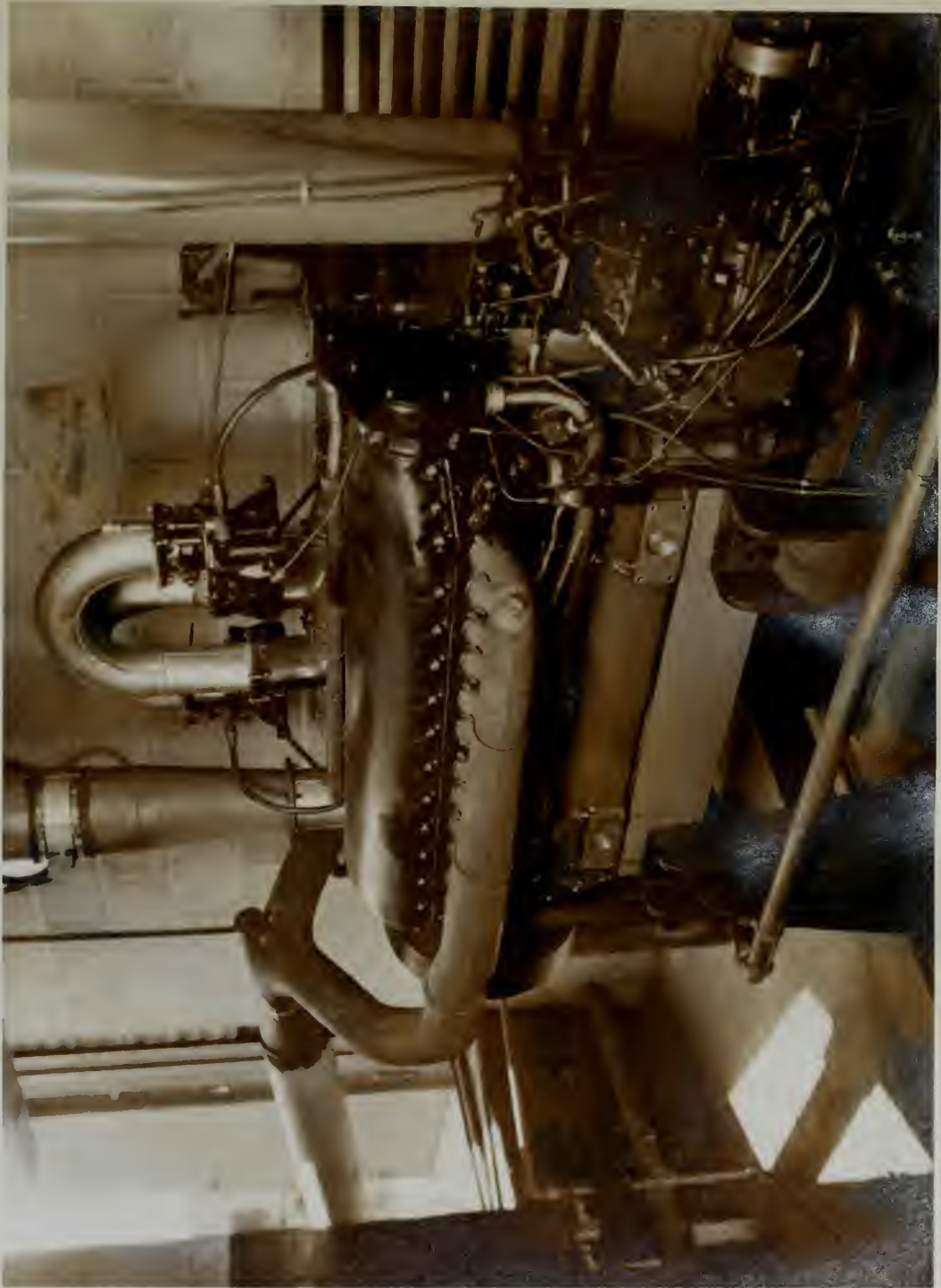




TEST SECTION MOUNTED ON BURNER





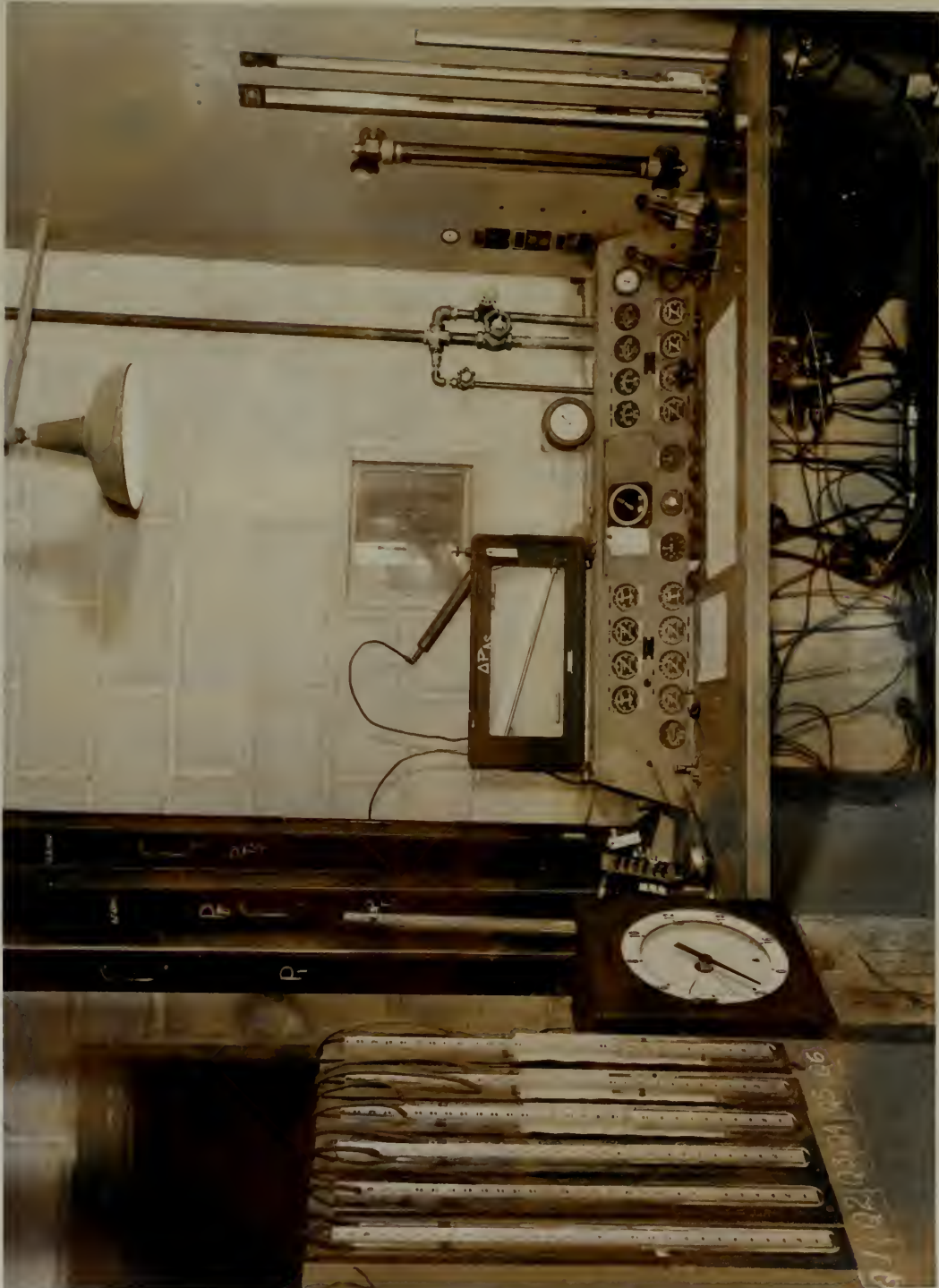


ANKER ENGINE

Fig. 8







CONTROL PANEL

Fig. 9





COOKING AIR METER

Fig. 10

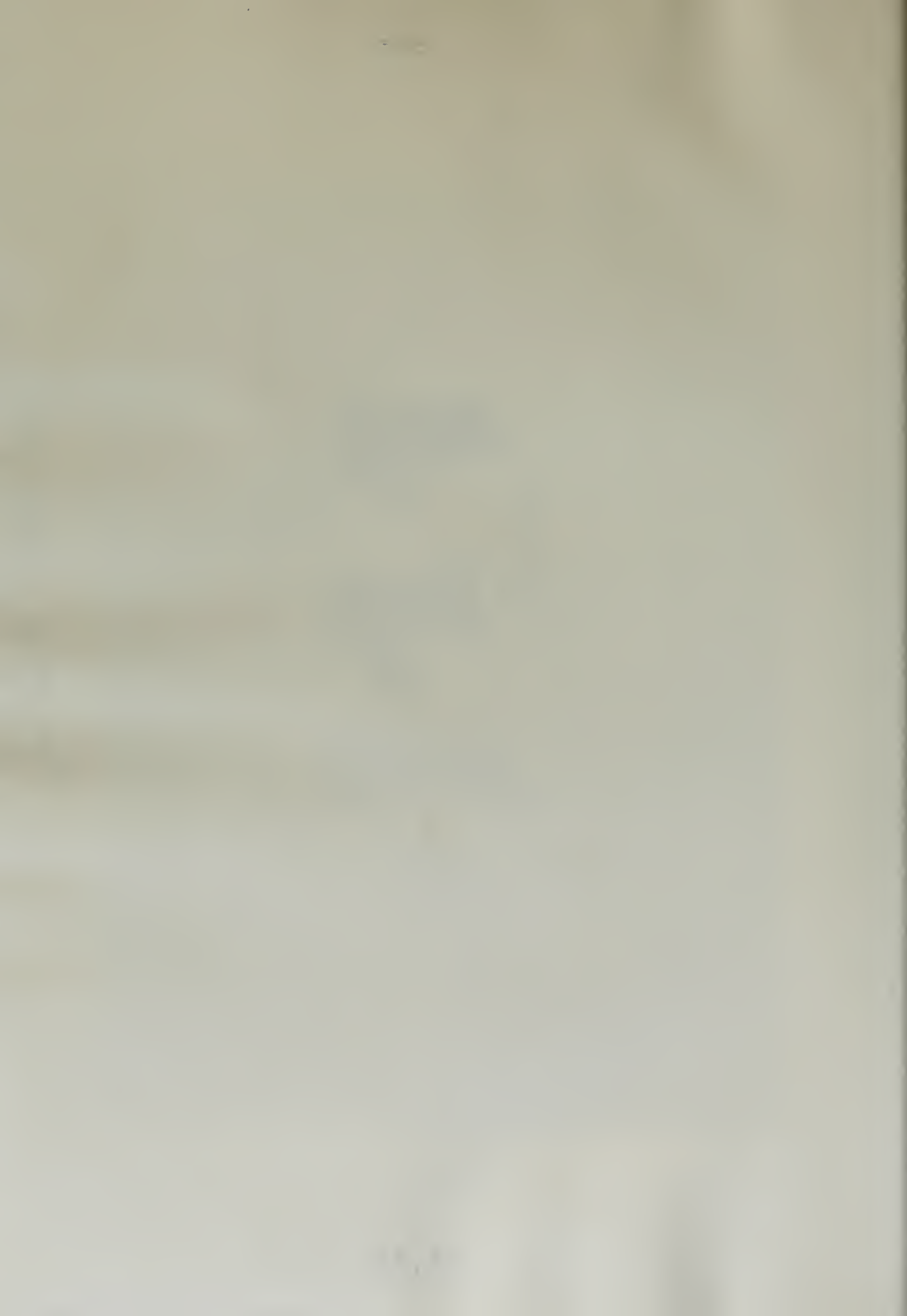




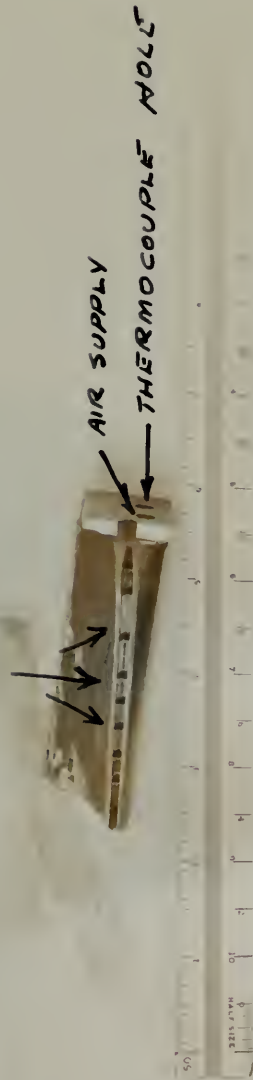


TEST BLADE (CONF. A)

Fig. 11



LEADING EDGE BLEEDS

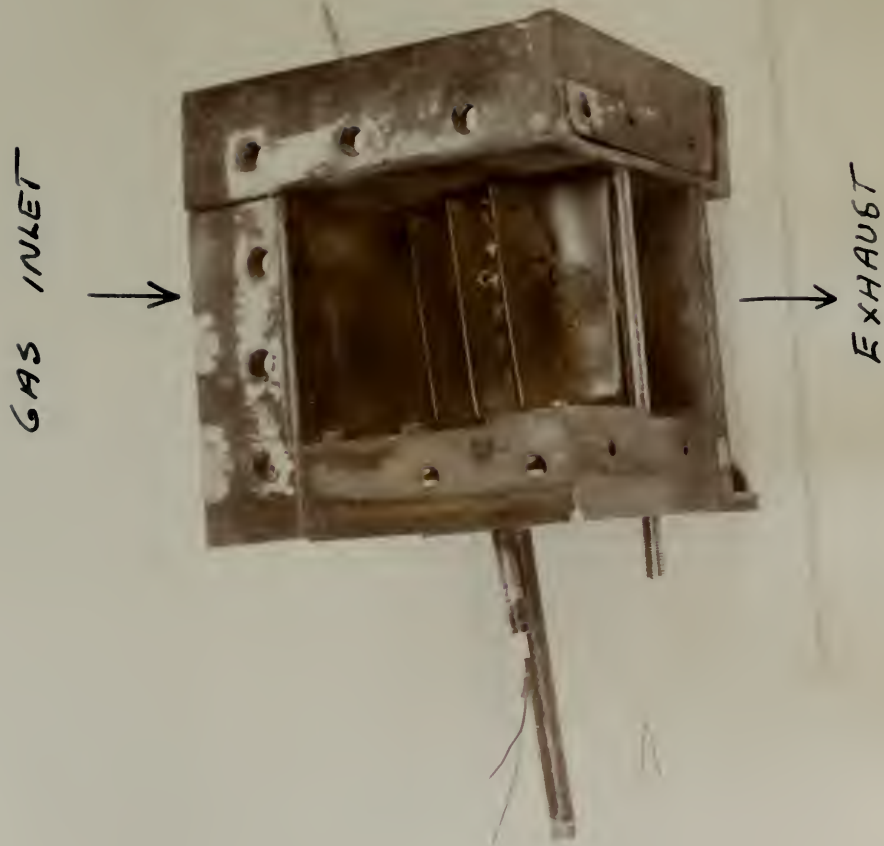


TEST BLADE (CONF. B)

Fig. 12







COMPLETE TEST SECTION

Fig. 13



- A- LEADING EDGE  
THERMOCOUPLE
- B- TRAILING EDGE  
THERMOCOUPLE
- C- COOLING AIR SUPPLY
- D- " " BLEEDS

-30-



MOUNTED TEST BLADE

Fig. 14





GAS INLET



EXHAUST



END VIEW OF BLADES

Fig. 15





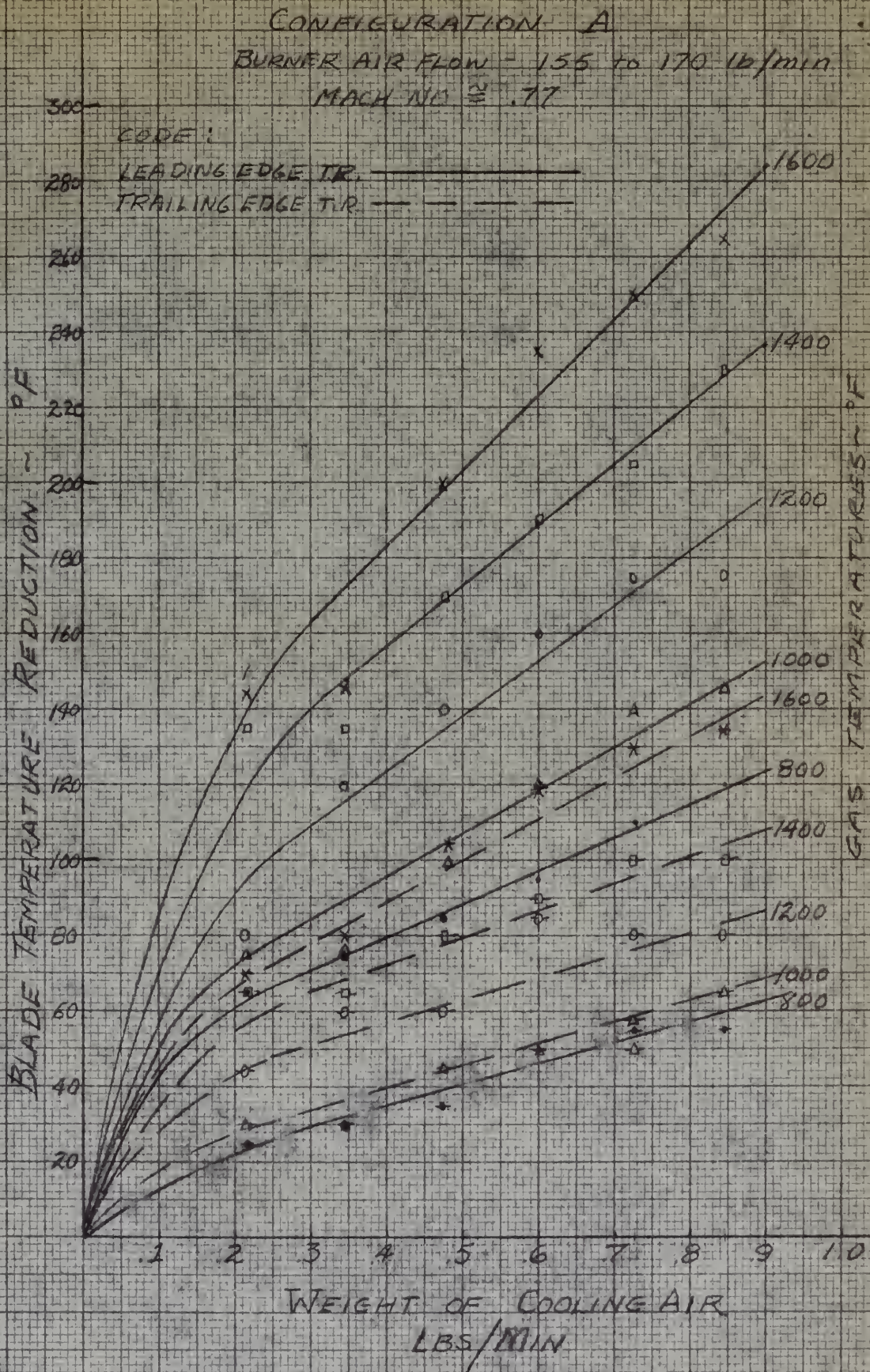
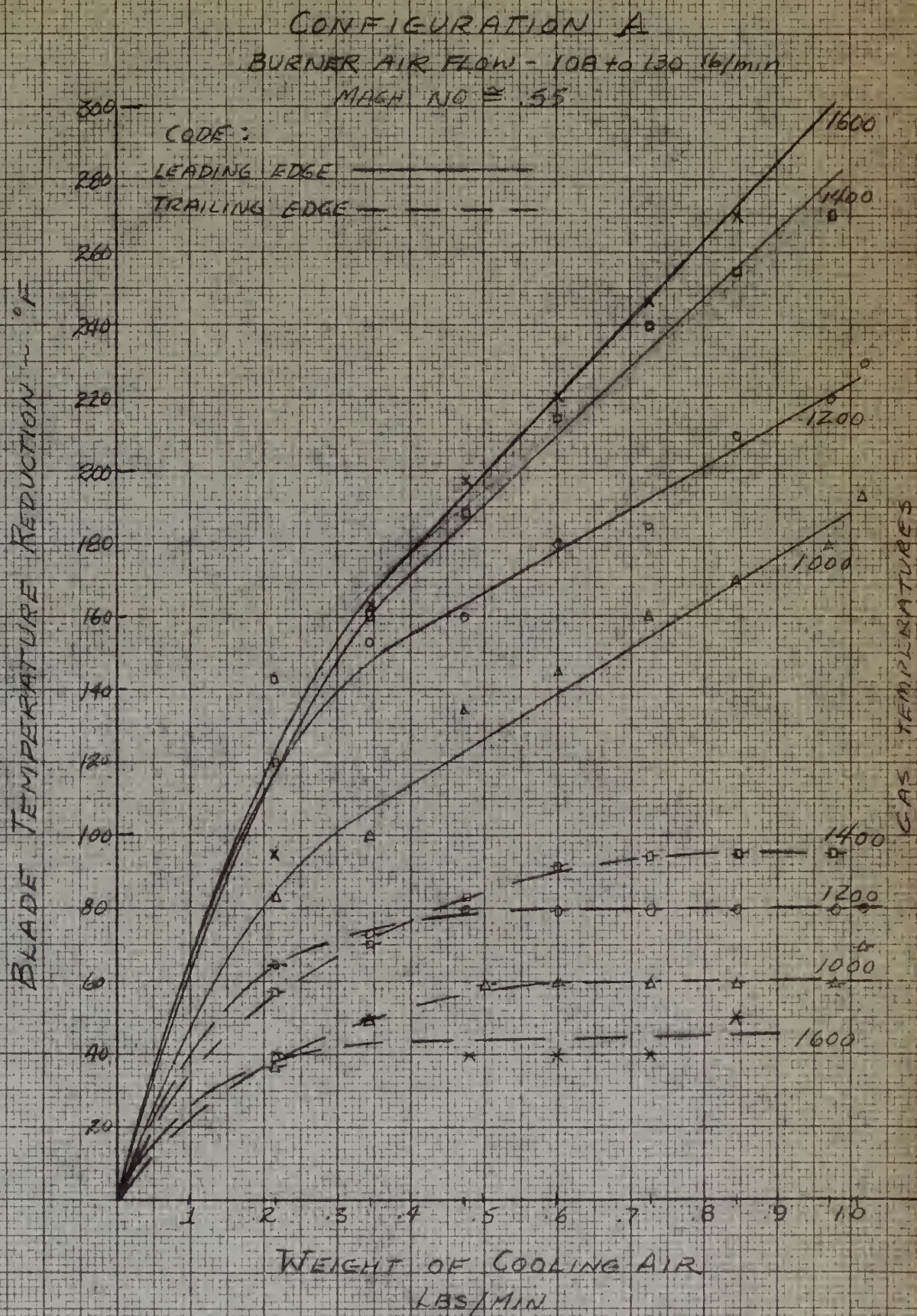


FIG 16









EFFECTIVENESS OF BOUNDARY LAYER  
 IN REDUCING BLADE TEMPERATURE

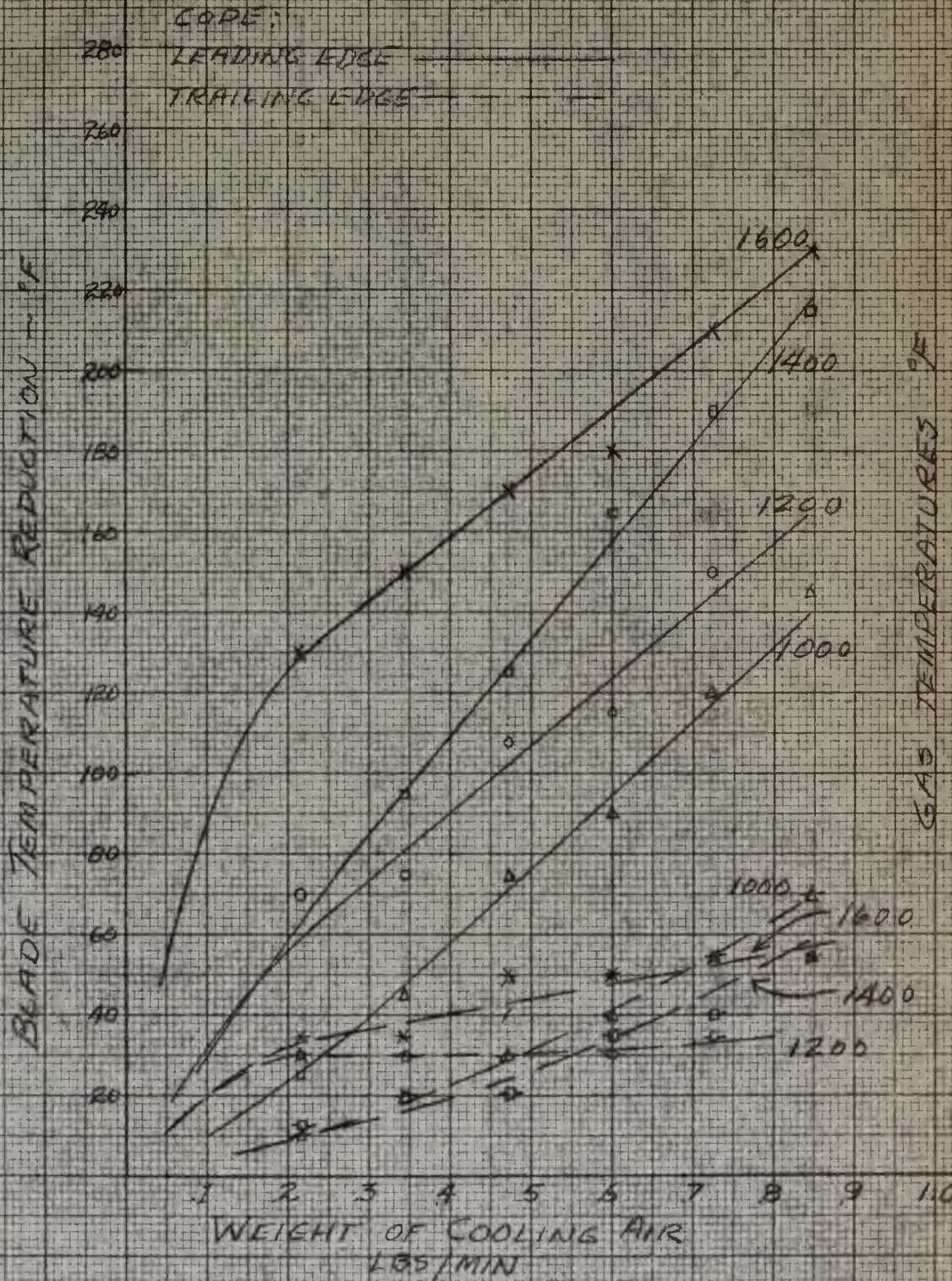




# CONFIGURATION B

BURNER AIR FLOW - 155-170 lb/min.

MACH NO  $\approx$  .77



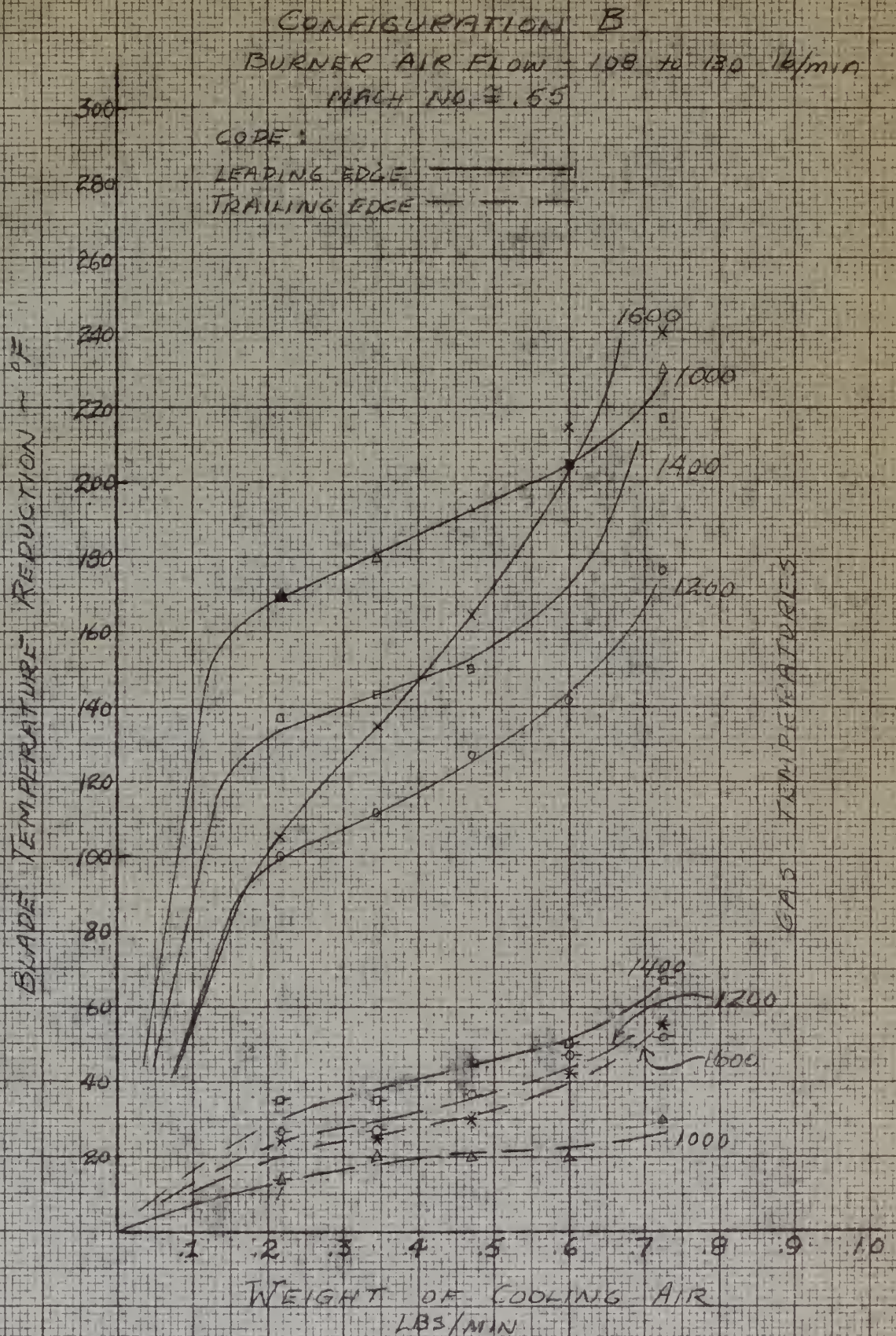
EFFECTIVENESS OF BOUNDARY LAYER  
IN REDUCING BLADE TEMPERATURES

Fig. 18



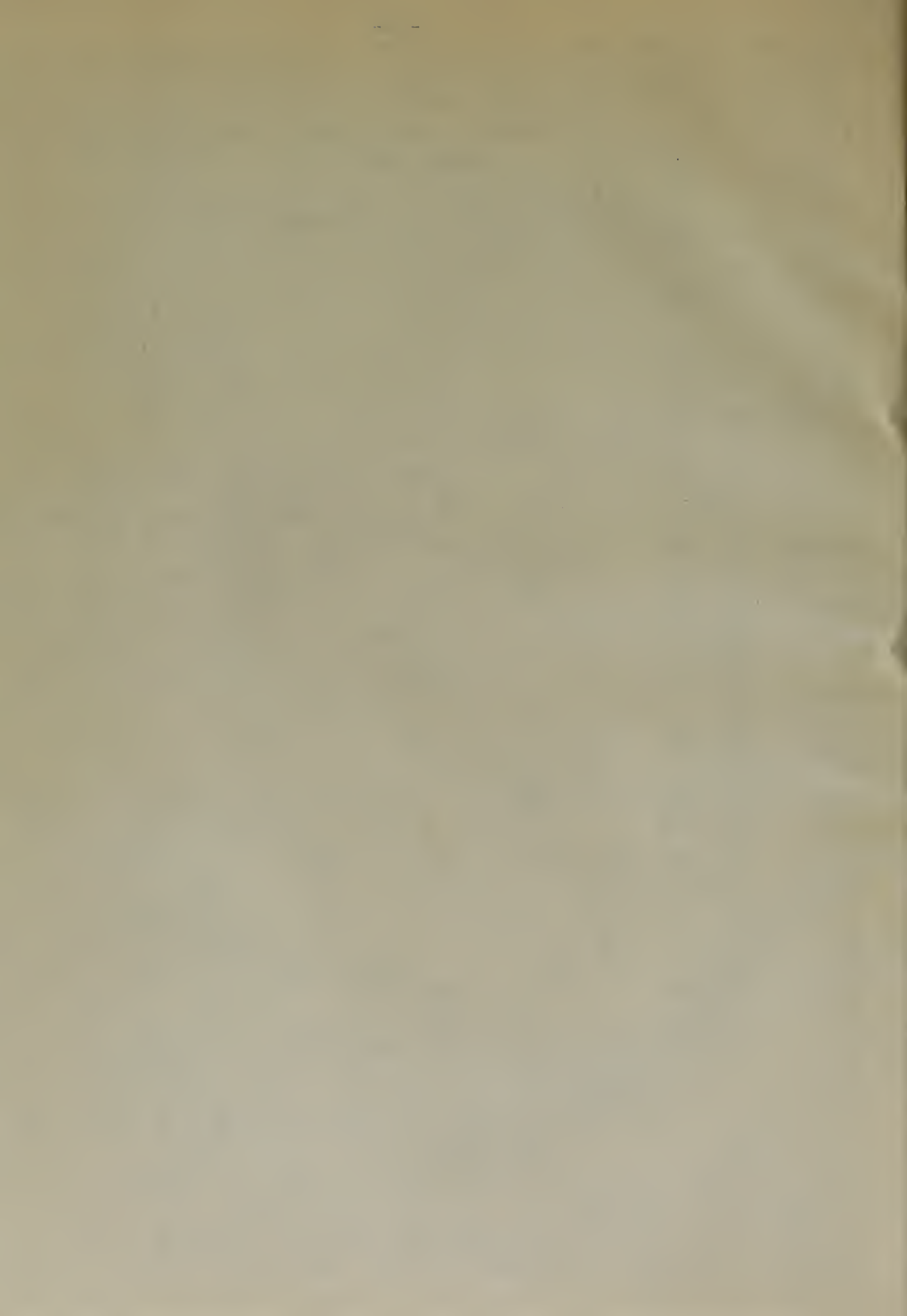






EFFECTIVENESS OF BOUNDARY LAYER  
 IN REDUCING BLADE TEMPERATURES

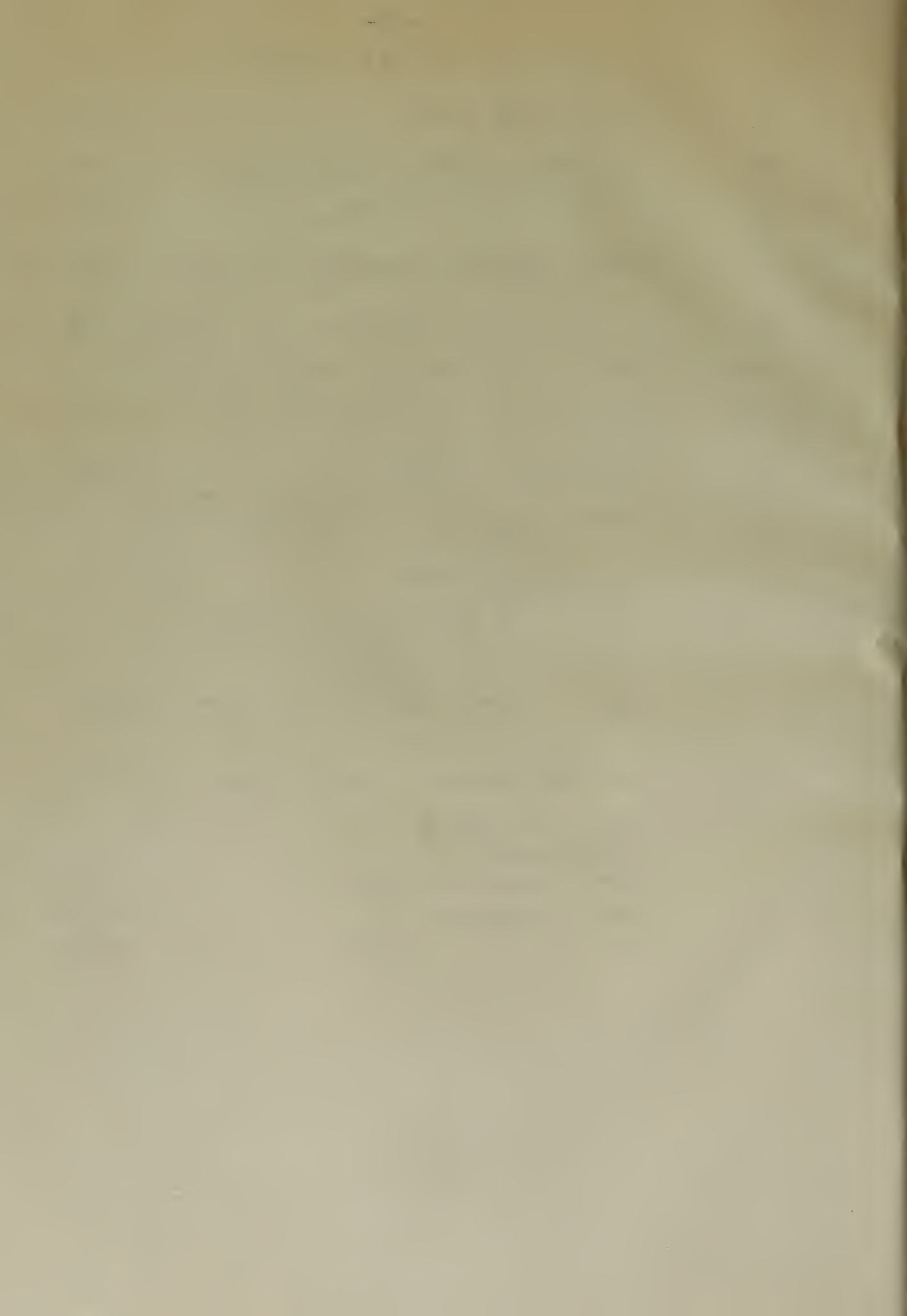
Fig. 19





# NOMENCLATURE

$P_t$	FUEL PRESSURE	psia
$P_3$	BURNER INLET STATIC PRESS	"Hg
$P_{t3}$	" " TOTAL "	"Hg
$P_4$	TEST SECTION STATIC "	"Hg
$P_{t4}$	" " TOTAL "	"Hg
$\Delta P_{CA}$	COOLING AIR ORIFICE PRESS DROP	"H <sub>2</sub> O
$\Delta P_{BA}$	BURNER " " "	"Hg
$q$	DYNAMIC PRESS	"Hg
$M_4$	TEST SECTION MACH NO.	
$T$	TEMPERATURE	°F
SUBSCRIPTS:		
	BLE BLADE LEADING EDGE	
	BTE " TRAILING "	
	M, R " ROOT	
	G, C GAS, AFFECTING HEAT TRANSFER	
T.R.	TEMPERATURE REDUCTION	°F
$W$	WEIGHT FLOW	
SUBSCRIPTS:		
	CA COOLING AIR	lb/min
	BA BURNER "	lb/sec
	f " FUEL	lb/hr





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- (2) Smith, G. O.; "Gas Turbines and Jet Propulsion for Aircraft," 4th Ed., Aircraft Books, Inc. 1946.
- (3) Stewart, R. W.; "Literature Survey on the Use of Ceramic Materials for Turbine Blading." Allis-Chalmers Manufacturing Co., Turbopower Development Department Report, 18 March, 1948.
- (4) McAdams, W. H.; "Heat Transmission;" 2nd Ed., McGraw-Hill Book Co., 1942.



# SAMPLE CALCULATIONS

Air metering: With standard sharp edged orifice configuration with flange taps.

Cooling air,

$$W_a = .8595 K D_2^2 \frac{(P_2 \times \Delta P)^{\frac{1}{2}}}{(T_a)^{\frac{1}{2}}} \quad (\text{ASME Power Test Code})$$

$T_a = 80^\circ \text{ F.}$  Air supply temperature

$D_2 = .75"$  Orifice diameter

$D_1 = 2.07"$  Pipe diameter

$K = .61$  From Fig. 34(a) of ASME Power Test Codes

$P_2 =$  Absolute outlet static pressure lb./in.<sup>2</sup>

$\Delta P =$  Orifice static pressure drop lb./in.<sup>2</sup>

$$W_a = \frac{.8596 \times .61 \times (.75)^2}{(540)^{\frac{1}{2}}} (P_2 \times \Delta P)^{\frac{1}{2}}$$

$$= .01268 (P_2 \times \Delta P)^{\frac{1}{2}}$$

P	P(psi)	P <sub>2</sub>	W <sub>a</sub> (lb./sec.)	W <sub>a</sub> (lb./min.)
.1	.0036	22.6	.00362	.217
.2	.0072	28.6	.00575	.345
.3	.0108	35.6	.00785	.471
.4	.0144	43.6	.01	.60
.5	.0180	50.6	.0121	.726
.6	.0216	57.6	.0141	.846
.7	.0252	64.6	.0162	.972
.8	.0288	71.6	.0182	1.09

Burner air,

$$W_{BA} = .8596 K \frac{D_2^2}{T_1^{\frac{1}{2}}} (P_2 \times \Delta P)^{\frac{1}{2}}$$

$K = .704$

$D_2 = 5.6$

Results are tabulated in Tables I and II.

SAFETY CALCULATIONS

With standard sharp edged orifice  
ventilation with linear case.

Flowing air.

$$Q = \frac{C_d A \sqrt{2 \Delta P}}{\rho}$$

(From Power Loss Table)

$Q = 100 \text{ m}^3/\text{s}$  Air supply requirement

$Q = 100 \text{ m}^3/\text{s}$  Orifice diameter

$Q = 100 \text{ m}^3/\text{s}$  Pipe diameter

$Q = 100 \text{ m}^3/\text{s}$  From Fig. 3.4 of ASHRAE Handbook  
Tables

$Q = 100 \text{ m}^3/\text{s}$  Standard orifice plate pressure  
loss

$Q = 100 \text{ m}^3/\text{s}$  Orifice plate pressure drop  
loss

$$Q = \frac{C_d A \sqrt{2 \Delta P}}{\rho}$$

Losses

$$Q = \frac{C_d A \sqrt{2 \Delta P}}{\rho}$$

$Q$	$\Delta P$	$Q$	$\Delta P$
100	1.0	100	1.0
100	1.0	100	1.0
100	1.0	100	1.0
100	1.0	100	1.0
100	1.0	100	1.0
100	1.0	100	1.0
100	1.0	100	1.0
100	1.0	100	1.0
100	1.0	100	1.0
100	1.0	100	1.0

Flowing air.

$$Q = \frac{C_d A \sqrt{2 \Delta P}}{\rho}$$

$Q = 100$   
 $Q = 100$

Results are tabulated in Tables 1 and 2.



Mach. number:

$$q = \frac{\gamma}{2} \rho V^2$$

$M_4$  = Test section mach. no.

$q$  = Test section dynamic pressure

$\gamma = 1.3$  for gas

$P_4$  = Test section static pressure

$$M_4 = \frac{2q}{\gamma P_4} = \frac{q}{.65P_4}$$

Results are tabulated in Tables I and II.





**DATE DUE**

27 MAR '5			
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[illegible]



The Thesis  
N4 N4

Ness

Boundary layer control  
as a method of gas tur-  
bine blade cooling.

11472

Thesis  
N4

Ness

Boundary layer control  
as a method of gas tur-  
bine blade cooling.

11472

thesN4

Boundary layer control as a method of ga



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